

Conversion of thermal energy into electricity via a water pump operating in Stirling engine cycle

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ABSTRACT

In this paper, the principle of heat energy conversion via Stirling pump into electricity is considered. New scheme of Stirling pump is proposed, that differs from known ones in application of offset heater and cooler and valves controlling the motion of liquid. The mathematical model is implemented to examine the liquid flow and gas heat exchange in cylinders and regenerator. The numerical simulation of engine's working cycle is conducted for the purpose of determining the characteristic parameters of its design. A possibility of achieving high thermal efficiency at acceptable power level is shown.

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1. Introduction

One of the new trends in the development of energy-saving technologies is the design of low-power and medium-power generating plants operating with bio-fuel, solar energy and other renewable energy sources. Use of such power plants is very important, in particular, in small settlements in the northern part of Russia and in the small islands of Aegean Sea. Attempts to create a universal heat engine operating with any kind of heat source, including household garbage or bio-fuel combustion and solar energy, bring the developers to the idea of the Stirling engine utilization. The thermodynamic cycle of Stirling heat engine is close to the Carnot's ideal one, that allows to convert heat of above-listed low-temperature heat sources efficiently. Other advantages of Stirling engine are long engine life and low noise level [1]. The main disadvantage of Stirling-type heat engine is its high cost that results from the complexity of heat supply and removal, the necessity of use of high heat-conducting gas (typically helium) as a working medium and the requirements of high-quality processing for many parts of engine.

The Stirling engine in which water columns play the role of pistons is proposed in 1971 [2] and is known as Fluidyne engine or Stirling pump. In Fluidyne engine, heat energy is first converted into hydraulic energy of oscillating water flow and then is converted into mechanical or electric energy. Within this general approach, several variants of Fluidyne engine scheme were

considered [3]. Being compared with a conventional Stirling engine, Fluidyne engine has less power per mass unit because, on one hand, the working cycle duration is much longer (about 0.1–1 s) and, on the other hand, the engine mass is higher due to water filling. There is conventional opinion that the efficiency of Fluidyne engine is low due to the fact that heat exchange between water and working gas doesn't allow operation at high temperatures. Nevertheless, in Fluidyne engine the only moving mechanical unit is the turbine (or other device converting hydraulic energy into electricity), the long cycle duration reduces requirements for heat exchange intensity and air may be used instead of helium as a working gas. This is why the cost of Fluidyne engine is less than that of conventional Stirling engine. Among the ideas proposed to develop a more efficient Fluidyne engine, it should be stressed the proposal to substitute water by a liquid metal that also can be used as a heat carrier. In this case, there is a possibility of achieving similar efficiency compared to conventional Stirling engine while hydraulic energy of liquid metal may be efficiently converted to electric energy in MHD generator [4]. This engine design is free of any solid moving part and the engine life can be extremely long.

In the present paper, a new type of Stirling engine with liquid piston is considered. The suggested scheme is first described in [5]. In the suggested scheme valves are used to control the process of gas and liquid motion. Due to valves system, the motion of liquid through turbine or other electricity generating unit is not oscillatory but single-direction, which simplifies the hydraulic energy conversion process. It is also supposed to use offset heat exchangers (see Fig. 1) instead of the usual heater and cooler. Furthermore, it is assumed that the working gas is taken from cylinder, is forced

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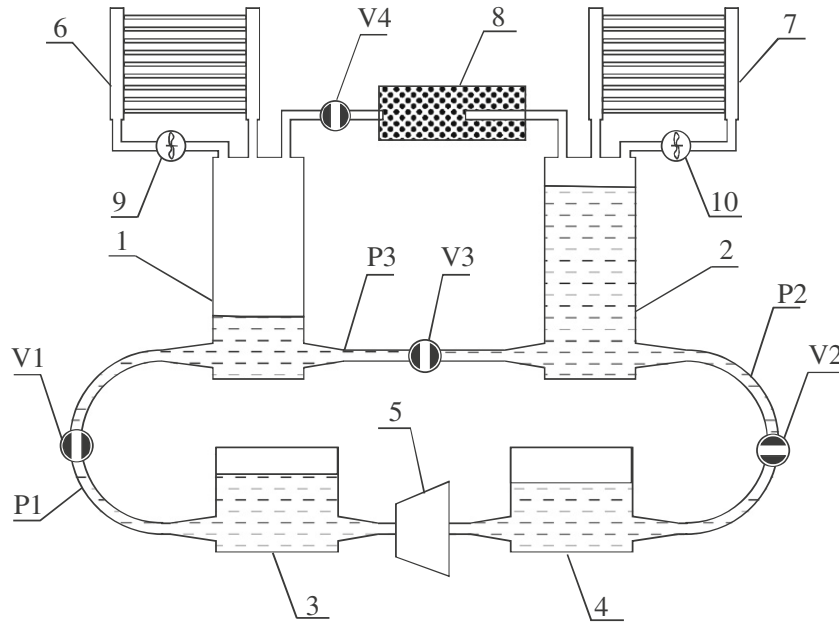


Fig. 1. Engine scheme. 1–heating cylinder; 2–cooling cylinder; 3–high-pressure tank; 4–low-pressure tank; 5–turbine; 6–hot heat exchanger; 7–cool heat exchanger; 8–regenerative heat exchanger; 9, 10–fans; V1–V4–valves; P1–P4–pipes.

to pass through offset heat exchanger by a fan and finally returned to the same cylinder. Such a method of heating and cooling is preferable when the cycle duration is long enough, because during single expansion or compression process the working gas passes through heat exchangers many times, thus the process is closer to isothermal. Heat-resistant floaters on the surface of water piston ensure heat insulation between working gas and water and prevent intensive evaporation of water.

2. General scheme of the Stirling pump and mathematical model of hydraulic and thermodynamic processes

The principle of Stirling pump may have various schemes of implementation and the search of the optimal one is out of the scope of the present study. According to the principle of Stirling engine, the working cycle must include a process where the gas in heating cylinder expands, approximately isothermally, producing work, a process where the gas in the cooling cylinder compresses, also approximately isothermally, consuming work and processes where the gas passes from one cylinder to the other through the regenerative heat exchanger. Using certain set of volumes and connecting pipes controlled by valves, these processes can be realised by means of liquid flowing into cylinders and out of them. The studied scheme (1) is probably the simplest one in which liquid water may flow through turbine in constant direction with approximately constant velocity. For this purpose two volumes, high- and low-pressure tanks, are included into the scheme. The liquid flows into the high-pressure tank during the gas expansion in the heating cylinder and flows out to the low-pressure tank during the process of gas compression in the cooling cylinder. Continuous flow through the turbine is provided by compression and expansion of gas, filling free spaces in the tanks. The volume of free spaces must be much more than the volume of water, passing through the turbine during each cycle. In this case pressure in tanks remains almost constant.

The mathematical model of processes in Stirling pump must describe the hydrodynamics of liquid flow in pipes and the gas dynamics in cylinders and regenerator. The approaches to the simulation of the processes in Stirling engine has developed during several decades starting from early zero-dimension models to

modern 3d models of gas flow in domains with moving boundaries. The task of simulation of Stirling heat pump includes interaction between the working gas and the liquid surface, thus a modern-level model must include a model of free surface liquid flow. This makes the model implementation rather demanding for calculation resources. As far as the aim of simulation is to determine a probable design of facility, many variants need to be calculated. Thus, a simpler model is preferable.

For the purpose of qualitative estimations zero-dimension models may be used to consider the hydrodynamics of liquid in pipes and heat exchange in cylinders. The work [6] is an example of using a model of such type for the simulation of Fluidyne engine. One-dimension model is the simplest approach for the description of gas flow and heat exchange in regenerator. One-dimensional models are known as rather rough approach for the simulation of conventional Stirling engine regenerators [7]. Nevertheless, it is reasonable to defer the complicated consideration until some principle points clarification takes place.

The viscous friction in liquid influences the efficiency of engine negatively, therefore pipes must be wide enough and have diffusers at their ends. The dynamics of liquid in a pipe with constant cross-section is described by the non-steady Bernoulli equation:

$$\rho l \frac{du}{dt} = p_{in} - p_{out} - \rho u_{out}^2 / 2 - (\xi_w + \xi_{in} + \xi_{out}) \rho u^2 / 2 \quad (1)$$

where ρ is the liquid density, u is hydraulic velocity, u_{out} is the hydraulic velocity at the end of diffuser part, l is the pipe length, p_{in}, p_{out} are pressures in volumes that are connected by given pipe, $\xi_w, \xi_{in}, \xi_{out}$ are friction factors relating to constant cross-section part of pipe, inlet and outlet diffusers respectively. The friction factors are calculated according to conventional empirical relations given in Ref. [8].

The gas state in the cylinders is described assuming uniformity of all the thermodynamic parameters of the gas in the given volume at the given time moment. Heat transfer rates in the heating and cooling cylinders' heat exchangers are found via the effective heat transfer coefficients:

$$W_{hcat} = H_{hcat}(T_1 - T_h), \quad W_{cool} = H_{cool}(T_2 - T_c) \quad (2)$$

where H stands for the effective heat transfer coefficients (watts per Kelvin degree of temperature difference), T_h , T_c are the gas temperatures in the cylinders, T_1 , T_2 are temperatures of the heat exchangers' metal walls. The temperatures of inner and outer sides of the metal walls are considered practically equal. The heat exchanger operates in the mode of non-steady regime; however, due to large heat capacity of metal walls, their temperature remains practically constant after the establishment of periodic operation mode. Thus, temperatures T_1 , T_2 in this calculation are taken constant and equal to 1000 and 300 K, respectively.

The equations of energy balance for the volumes are constituted by the cylinders and the heat exchangers connected to them and include items describing the inflow of thermal energy in the heat exchangers, gas expansion or compression work and the enthalpy inflow or outflow with the additional mass supplied from the regenerator. The corresponding equations are as follows:

$$\frac{1}{\gamma-1} \frac{d}{dt} (pV) = W_{\text{heat,cool}} - p \frac{dV}{dt} + \frac{\gamma}{\gamma-1} RT^* \frac{dv}{dt} \quad (3)$$

where p , V are pressure and volume of gas in the given cylinder respectively and γ is the adiabatic exponent. When the air valve is opened, the temperature T^* is equal to the gas temperature in the cylinder when there is outflow of the gas or equal to the gas temperature on the regenerative heat exchanger outlet when the gas is supplied into the cylinder. In other processes the amount of substance in the cylinders does not change and the item in (Eq. (3)) containing T^* , is absent.

The regenerator matrix is considered as N layers through which the gas passes consequently. The gas temperature on the outlet of the i th layer will further be designated as T_i and the temperature of the matrix layer as T_i^f ; the layers will be numbered in the order of gas passing through them. Intensity of heat exchange between the gas and a matrix layer can be expressed through the gas temperature on the layer outlet. In this case, the energy conservation equation will be as follows:

$$\frac{\gamma R}{\gamma-1} \dot{v} (T_i - T_{i-1}) = \alpha_i S_L (T_i^f - T_i) \quad (4a)$$

where $\dot{v}$ is the quantity of substance passing through regenerator per time unit (mole/s), α_i is heat transfer coefficient depending on the gas velocity, density and temperature, S_L is the metal–gas contact surface area for single layer. The heat transfer coefficient is calculated through empirical relation for random fiber regenerator matrix [9]. Heat exchange between gas and matrix layer may also be expressed through the average gas temperature on the given layer, thus giving the following expression:

$$\frac{\gamma R}{\gamma-1} (T_i - T_{i-1}) = \alpha_i S_L (T_i^f - (T_i + T_{i-1})/2) \quad (4b)$$

Option (Eq. (4a)) gives the following expression for the temperature on the layer outlet:

$$T_i \left(\alpha_i S_L T_i^f + \frac{\gamma R}{\gamma-1} \frac{dv}{dt} T_{i-1} \right) / \left(\alpha_i S_L + \frac{\gamma R}{\gamma-1} \frac{dv}{dt} \right) \quad (5a)$$

while option (Eq. (4b)) results in

$$T_i \left(\alpha_i S_L T_i^f + \left(\frac{\gamma R}{\gamma-1} \frac{dv}{dt} T_{i-1} - \frac{\alpha_i S_L}{2} T_{i-1} \right) \right) / \left(\frac{\alpha_i S_L}{2} + \frac{\gamma R}{\gamma-1} \frac{dv}{dt} \right) \quad (5b)$$

Using of (Eq. (5b)) gives smaller error at low heat transfer intensity:

$$\alpha_i S_L \ll \frac{\gamma R}{\gamma-1} \frac{dv}{dt},$$

However, if the above condition is unfulfilled, then instability occurs. On the contrary (Eq. (5a)), gives physically correct result for any conditions.

In the present study the heat conduction in the regenerator matrix is neglected. Thus, energy conservation law for the matrix gives the following equation for the layers' temperature:

$$c_L \frac{dT_i^f}{dt} = \frac{\gamma R}{\gamma-1} \frac{dv}{dt} (T_{i-1} - T_i) \quad (6)$$

c_L is heat capacity of matrix layer. Flow of the gas through the regenerator is defined by processes in the hydraulic system pumping the liquid into one cylinder and out of the other one, as well as by the rates of heating and cooling in the cylinders. The resistance of regenerator is supposed to be negligible, which must be verified after the estimation of engine regime parameters. Under this assumption, the pressures in the cylinders are equal:

$$P_{\text{heat}} = P_{\text{cool}}. \quad (7)$$

Considering that gas flow dv/dt set at a given value, the (Eq. 4 and 5) allows to determine the increase of the thermodynamic parameters on a small time interval. The increase of pressures in the cylinders will depend on the selected value of the flow. In this case, they must satisfy the condition described by (Eq. (7)):

$$\Delta p_{\text{heat}} \left(\frac{dv}{dt} \right) = \Delta p_{\text{cool}} \left(\frac{dv}{dt} \right). \quad (8)$$

Considering (Eq. (8)) as the equation recorded for the gas flow and solving it numerically, then the flow, the increase of thermodynamic parameters of the gas and, according to (Eq. (6)), the increase of temperatures of the regenerator layers can be determined.

3. Working cycle of Stirling pump

The working cycle of considered scheme of Stirling pump is guided by opening and closing a set of valves. The sequence of valve operations may be realised via an external programmed controller. The control program must react in events such as change of flow direction and pressure equalization. The optimal sequence of these operations is found after a number of numerical simulations of the facility hydraulics and thermodynamics. As a result of the calculations, the working cycle of the engine is determined and consists of the following processes (see Fig. 2):

Process 1. As valve V1 is opened and the rest are closed, the gas in heating cylinder 1 performs the expansion process and forces the liquid through the pipe P1 into the high-pressure tank 3. During the process, the gas pressure in cylinder becomes lower than the pressure in tank, and the liquid in pipe P1 begins braking. At the moment of its stop, valve V1 is closed and process 1 finishes. By the end of this process, the pressure in the cooling cylinder 2 is higher than that in the heating cylinder.

Process 2a. After the opening of valve V3, water in the pipe P3 flows with acceleration to the heating cylinder. The gas is compressed in the heating cylinder and is expanded in the cooling cylinder, thus, the pressures in the cylinders become equal after certain time interval. At this moment valve V4 opens.

Process 2b. The gas starts to flow from the heating cylinder into the cooling one through the regenerative heat exchanger 8. There the gas gives its heat to the matrix. Since the overall gas volume is preserved, the gas pressure decreases in both cylinders. Water continues flowing into the heating cylinder braking weakly due to friction. When the volume of gas in the heating cylinder becomes lower than a certain value, valve 8 closes.

Process 2c. The water still flow through pipe P3, so gas volume in heating cylinder decreases and volume in cooling cylinder increases. The pressure difference between cylinders grows and water in the pipe begins braking. Valve V3 closes when the velocity of water falls down to zero. In the same time valve V2 opens.

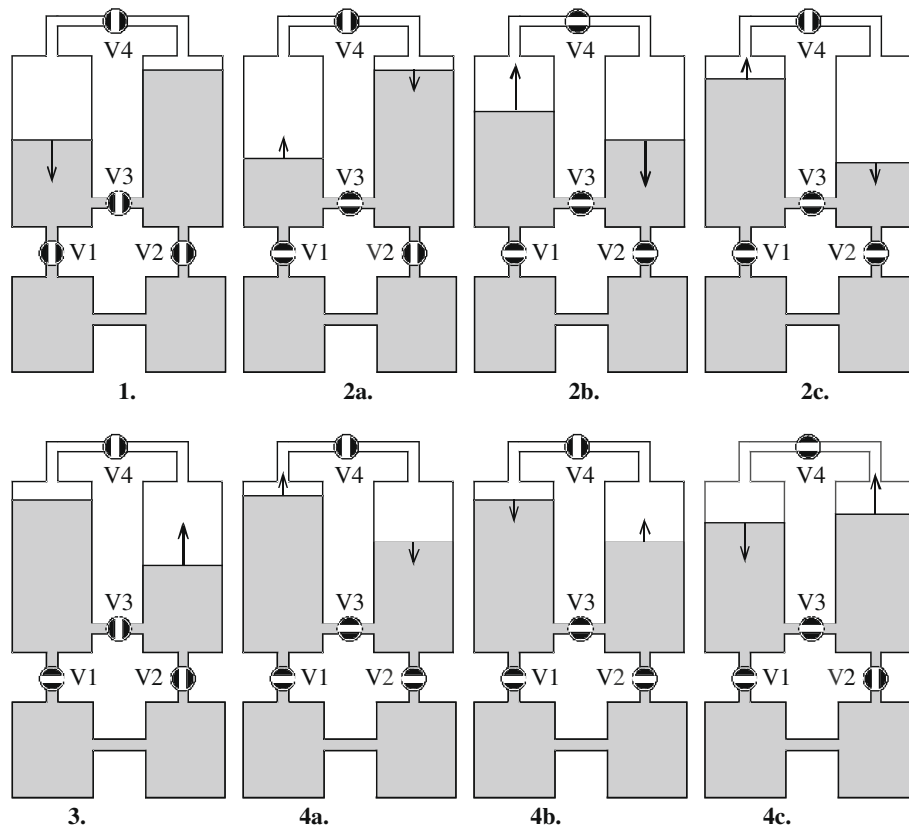


Fig. 2. Working cycle of engine; simplified scheme. 1–4 – stages of cycle.

Process 3. The gas in cooling cylinder 2 is compressed in the process close to isothermal one by the liquid supplied through the pipe P2 from the low-pressure tank 4. The process is finished when the water in the pipe P2 is braked by increasing gas pressure to zero velocity and valve V2 is closed. By the end of this process, the pressure in the heating cylinder is greater than that in the cooling cylinder.

Process 4a. After the opening of valve V3, water in the pipe P3 flows with acceleration to the heating cylinder. The gas compresses in the heating cylinder and expands in the cooling cylinder, thus, after certain time the pressures difference between cylinders changes its sign. As a result, water in P3 stops and begins to flow to the cooling cylinder.

Process 4b. The gas compresses in the cooling cylinder and expands in the heating cylinder, thus, the pressures in the cylinders become equal after certain time interval. At this moment valve V4 opens.

Process 4c. The process is the inverse of process 2b. The process finishes when the volume of gas in the cooling cylinder becomes lower than a certain value, and valve V4 closes.

Process 4d. (not shown in Fig. 2). Water flows to the cooling cylinder until stopping due to rising pressure drop. At the moment of flow stopping, valve V3 closes, valve V1 opens and cycle starts again.

During processes 1 and 3 gas cools due to expansion and heats due to compression, but at the same time fans forces it to flow through the heat exchangers, where it receives heat from a heat source and gives heat to cooler. If gas flow through heat exchangers is high enough, the gas temperature in these processes remains approximately constant.

Between the high-pressure and low-pressure tanks, hydroturbine 5 is mounted. Water passes through it permanently and with

practically constant flow rate. The turbine rotates the rotor of the electric generator.

The dynamics of liquid in pipes is of oscillatory type, like in Fluidyne engines, but under the control of valves the liquid motion passes only part of oscillation period. In the processes 1 and 3 it passes a half of period, in 2a, 2c, 4c – a quarter of period, in 4a – three quarters. Qualitative analysis shows that the period of oscillation is proportional to square root of pipe length divided by pipe cross-section. In the same time friction losses are proportional to the square of velocity amplitude. This way, proportional increase of pipe length and cross-sections reduces friction losses keeping thermodynamic cycle parameters approximately constant. In the same time, mass and length characteristics of engine increase.

During the numerical simulation of the engine processes, parametric analysis was performed in order to achieve an acceptable thermal efficiency and engine mass. The optimal structural parameters of the scheme were calculated and are given in Table 1.

The pipes are connected to the volumes via the diffusers with expansion ratio of 3.

The characteristic volumes and pressures in the cylinders and tanks were determined as a result of simulation and appeared in Table 2.

At the above-listed engine parameters, the maximum air velocity in the free cross-section of regenerator was about 5 m/s and the regenerator resistance was less than 0.2 bar. This is lower than typical pressure difference between volumes, thus the resistance of regenerator doesn't affect much the gas thermodynamics. Besides, the resistance of pipe 3 during processes 2b and 4c is 1–2 bars, thus the braking of water flowing in pipe 3 occurs mainly under the influence of pipe hydraulic resistance. Pressure difference between cylinders produced by regenerator resistance doesn't influence much this process.

Table 1
Optimal structural parameters.

Stirling pmp parameter	Value
Working gas	Air
Amount of substance of the working gas	144 mol
Regenerator matrix	Random wire
Wire diameter	0.2 mm
Wire material	Stainless steel
Porosity	0.7
Regenerator matrix cross-section	0.1 m ²
Regenerator matrix length	0.2 m
Volumes of the offset heat exchangers	20 litres each
Gas flow rate through the offset heat exchangers	10 kg/s
<i>Pipe' parameters:</i>	
Heating cylinder – high-pressure tank	3 m, Ø 0.1 m
Cooling cylinder – low-pressure tank	3 m, Ø 0.1 m
Heating cylinder – cooling cylinder	1 m, Ø 0.3 m
Walls' roughness	0.1 mm

Table 2
Characteristic volumes and pressures in Stirling pump cylinders and tanks.

Stirling pump parameter	Value
Maximum volume of gas in the heating cylinder (the heat exchanger is taken into account)	85 l
Maximum volume of gas in the cooling cylinder (the heat exchanger is taken into account)	80 l
Maximum pressure in the heating cylinder	95 bar
Maximum pressure in the cooling cylinder	100 bar
<i>Pressures in the accumulating tanks:</i>	
High-pressure tank	78 bar
Low-pressure tank	48 bar
<i>Working gas temperature range:</i>	
In the heating cylinder	760–50 K
In the cooling cylinder	330–20 K
Cycle duration	0.54 s
Thermal power	266 kW
Electric power	97 kW
Thermal efficiency	0.37
Power consumption of fans drive	5–0 kW

It is easy to estimate conductive heat flux through the regenerator matrix. Taking under consideration the size, temperature difference, material and structure of matrix, the heat flux is estimated to approximately 2 kW, which is much less than thermal power of engine. This leads to the conclusion that thermal conductivity of regenerator matrix doesn't influence much the engine efficiency.

The gas parameters' time dependencies in all the thermodynamic processes of the given engine's cycle, found by numerical simulation in the optimal mode, are given in Fig. 3–7.

In Fig. 3, results of the simulation are presented as a thermodynamic cycle in the coordinates of mean gas pressure and mean specific volume. The mean gas pressure is defined as $\sum PV / \sum V$ for two cylinders and the mean specific volume is defined as $\sum V / \sum m$. This cycle is close to the generalized regenerative Carnot one. For comparison, ideal isotherms are indicated by dashed lines in the diagram.

In Fig. 4, dynamics of gas volumes of the heating and cooling cylinders is presented. Time moments 1, 2, 3 and 4 correspond to the node points of the cycle on Fig. 3. In the diagrams, volumes are constant during the time intervals in which the liquid in the corresponding cylinder is still. In this case, the gas volume is fixed on the level of 20 l i.e. on the internal volume of the offset heat exchangers. At this moment, in the other cylinder isothermal process occurs. The isochoric process of the gas displacement from the heating cylinder into the cooling one occurs during the interval 2–3 where the sum of gas volumes is not changed.

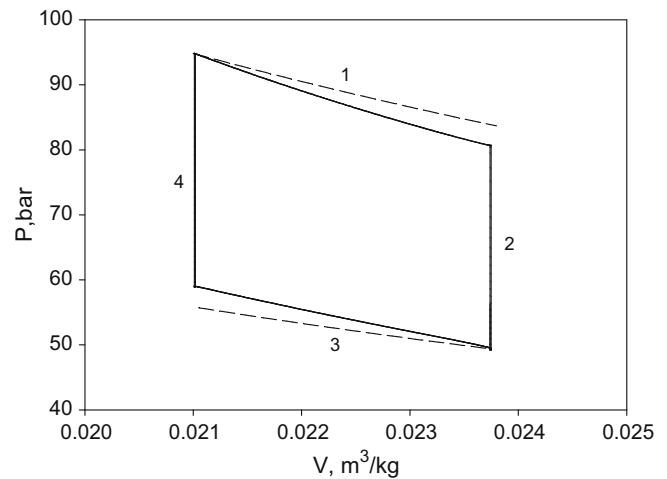


Fig. 3. Thermodynamic cycle. Numbers on plot corresponds the numbers of process. Dashed lined are ideal isotherms.

In Fig. 5, dynamics of pressures in the heating and cooling cylinders is presented. The process of isothermal expansion in the heating cylinder corresponds to time interval 1–2. In this case, the isochoric cooling with pressure decrease occurs in the cooling cylinder where the gas is concentrated in the offset cooler volume. Further on, the process of isochoric cooling of the gas in the regenerator, common for both cylinders, occurs, and, respectively, pressure in the cylinders decreases to the pressure of about 45 bar. Time interval 3–4 corresponds to isothermal gas cooling in the cooling cylinder and isochoric heating of the gas in the offset heat exchanger of heat cylinder.

Fig. 6, as compared to Fig. 3 and 4, provides additional information on the conformity of processes 1–2 and 3–4 to isothermal heating and cooling. As it can be seen in the diagrams, temperature variation is about 60 K. Average temperature of the gas in process 1–2 constitutes 850 K (150 K lower than that of the wall in the offset heat exchanger), and in process 3–4 – approximately 350 K, i.e. 50 K higher than temperature of the wall cooled by water in the cooler.

In Fig. 7, temporal dependencies of velocity of the liquid in the pipes are shown. The first and the second diagrams indicate situation in the pipes connecting the cylinders with the corresponding tanks. These diagrams are quite obvious, since they reflect simple dependency: half-period of the oscillation process beginning with opening of the valve and ending with closing of the latter. The third diagram demonstrates the dynamics of liquid flow velocity in the pipe connecting the cylinders. Non-zero velocity values correspond to the isochoric processes in which the liquid flows from one cylinder into the other. Rapid increase of the velocity in these processes is explained by a rather high-pressure drop and small mass of the liquid in the connecting pipe. Stabilisation of the velocity occurs at the moment of opening of the air valve equalizing pressure in the cylinders, after which velocity absolute value decreases slowly due to friction. Sharp decrease of velocity absolute value occurs after closing the air valve in the end of isochoric processes.

It should be noted that the maximum velocity value is about 25 m/s. It is a very high velocity which would result in significant hydrodynamic losses due to friction. Apparently, this is the reason for the real value of the efficiency of 37% to turn to be significantly lower than the ideal value defined for the regenerative Carnot cycle: heat source temperature being 1000 K and cooler temperature being 300 K, Carnot efficiency would be 70%. In fact, the real efficiency will be even lower, since contemporary hydroturbines have

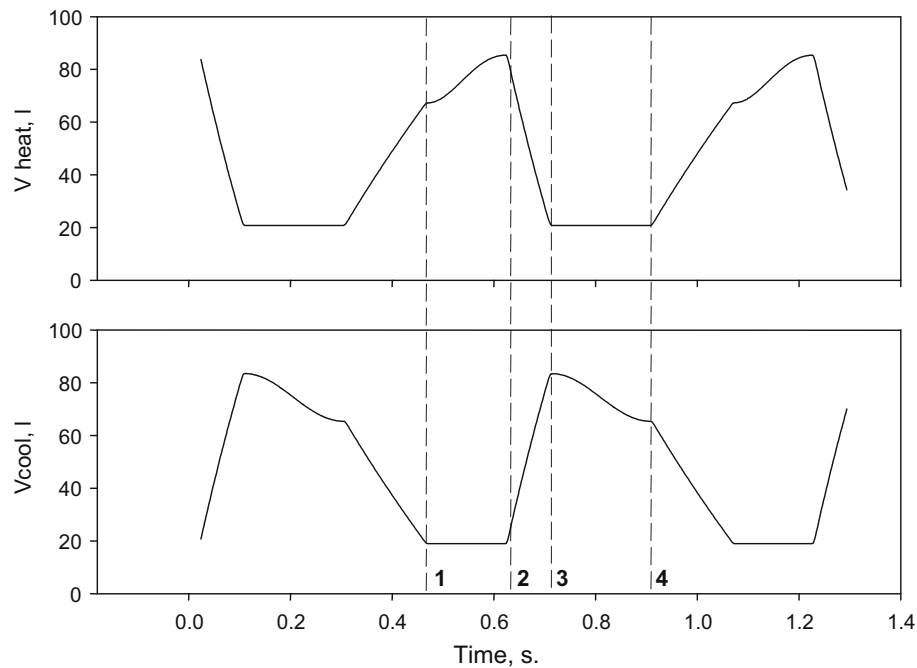


Fig. 4. Time dependency of volumes of the heating and cooling cylinders. Marked are the time moments corresponding to the cycle nodes.

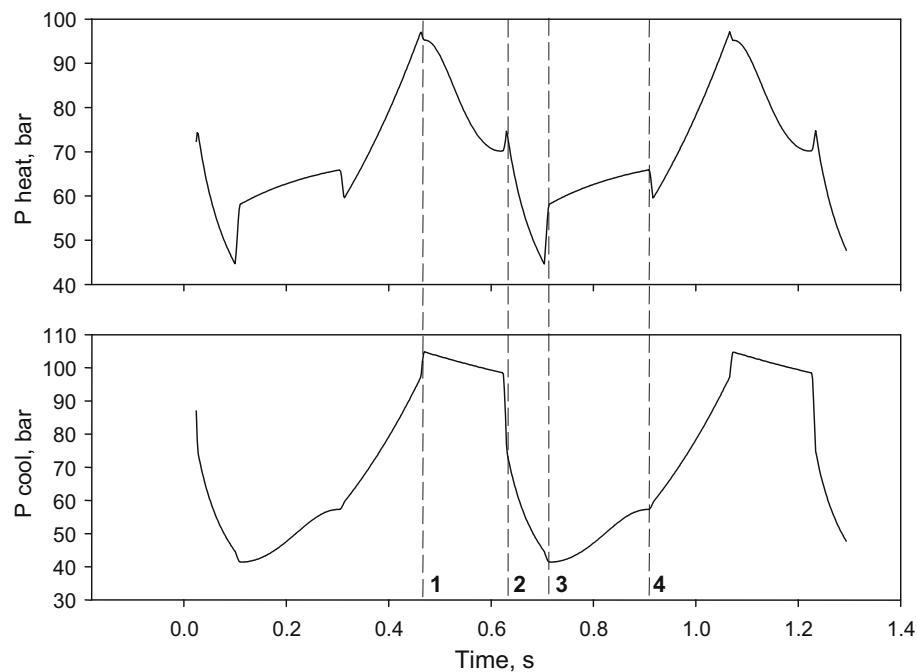


Fig. 5. Time dependency of pressures in the heating and cooling cylinders.

efficiency of 90%. Respectively, the real efficiency of the water pump converting thermal energy into electricity should be calculated as $0.37 \times 0.9 = 33\%$.

4. Discussion

The principle of Stirling pump operation is used to design a cheaper Stirling engine. Numerical analysis of hydraulic and heat exchange processes has been conducted for the variant of Stirling pump proposed in this paper. The calculations have shown that

high efficiency can be achieved with the use of air as a working gas. There is no solid moving part in engine based on Stirling pump, except the turbine and the electric generator rotor. The proposed Stirling pump shows long cycle duration which gives a possibility to intensify the useful heat exchange processes in cylinder with the help of offset heat exchangers and to substitute helium by air, nitrogen or other cheap gas. Besides, longer cycle duration means less strict requirements for the quality of regenerator matrix. Implementation of offset heat exchangers and regenerator allows achieving thermal efficiency higher than 30% at the heater

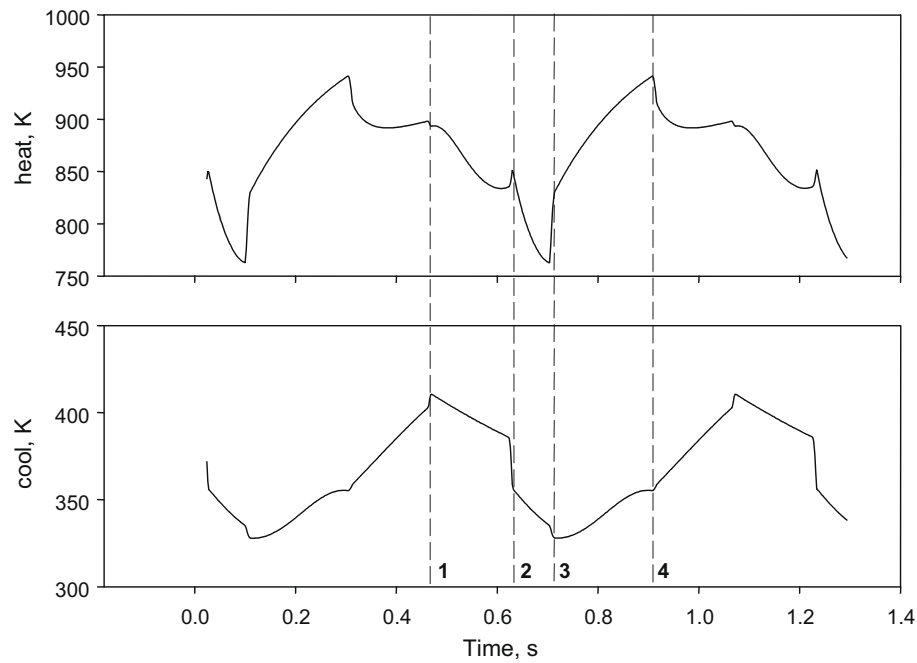


Fig. 6. Time dependency of temperatures in the heating and cooling cylinders.

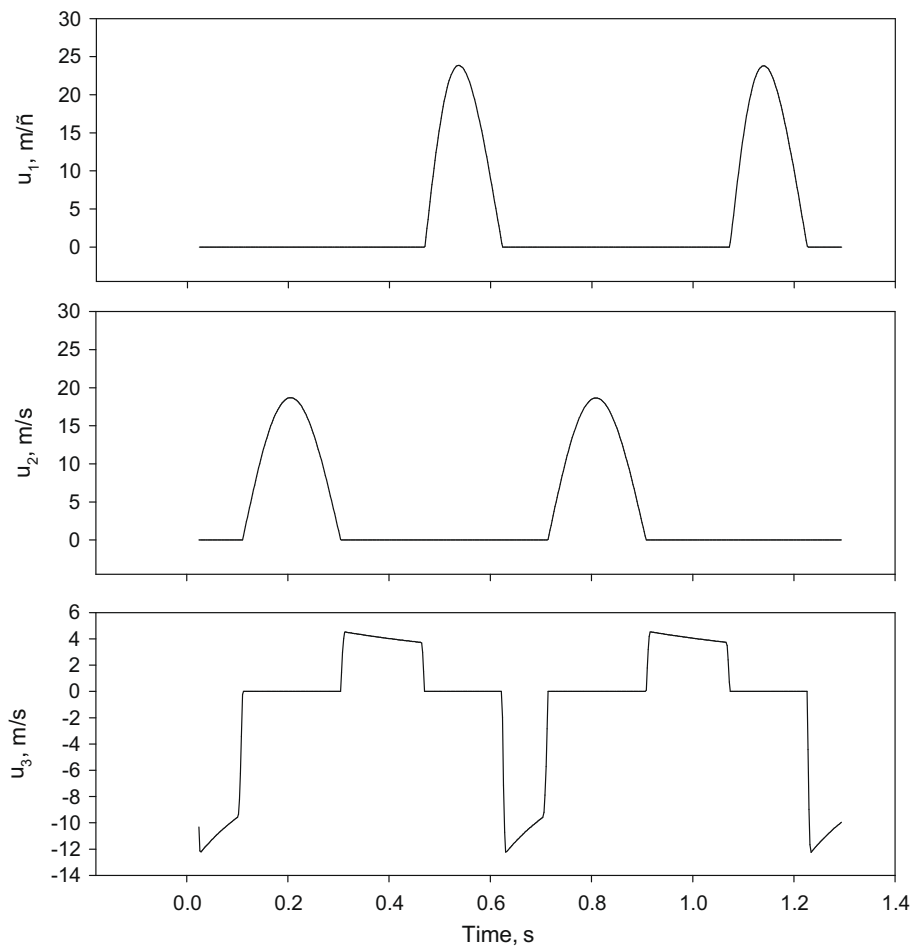


Fig. 7. Velocities in the pipes: u1 – pipe P1, u2 – pipe P2, u3 – pipe P3.

and cooler temperatures, typical for conventional Stirling engines. The friction losses in water pipes are significant factors limiting the power and efficiency of Stirling pump. The use of valves controlling the water flow in pipes allows to produce unidirectional flow of liquid through turbine. This feature is very useful for the inclusion of Stirling pump into pumped-hydrostorage systems.

However, the realization of the proposed scheme is debatable due to some difficulties related to its construction. One of the difficulties is the strict requirements for valves related to short actuation period and low hydraulic resistance in open state. Another practical problem is the realization of components sealing such as hydroturbine and fans. Also, more detailed studies are necessary to clarify the influence of water evaporation and temperature non-uniformity in cylinders.

Nevertheless, it can be concluded that the energy conversion based on the principle of proposed Stirling pump is a promising area.

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